

THEORETICAL VERSUS EXPERIMENTAL RESULTS FOR THE ROTORDYNAMIC  
COEFFICIENTS OF ECCENTRIC, SMOOTH, GAS ANNULAR  
SEAL ANNULAR GAS SEALS

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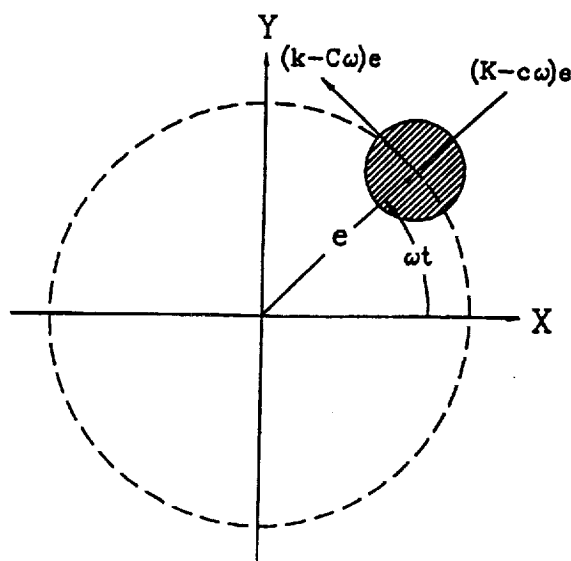
INTRODUCTION

ANNULAR GAS SEAL MODEL

- Annular gas seal exhibiting small motion about a centered position

$$-\begin{Bmatrix} F_x \\ F_y \end{Bmatrix} = \begin{bmatrix} K & k \\ -k & K \end{bmatrix} \begin{Bmatrix} X \\ Y \end{Bmatrix} + \begin{bmatrix} C & c \\ -c & C \end{bmatrix} \begin{Bmatrix} \dot{X} \\ \dot{Y} \end{Bmatrix}$$

- Rotordynamic force components acting on a rotor



Forces on a whirling rotor

## TEST APPARATUS

- Rotor shaft / Pivot shaft arrangement
- Horizontal excitation through shaker head arrangement
- Load cell / Accelerometer arrangement
- Cross sectional view

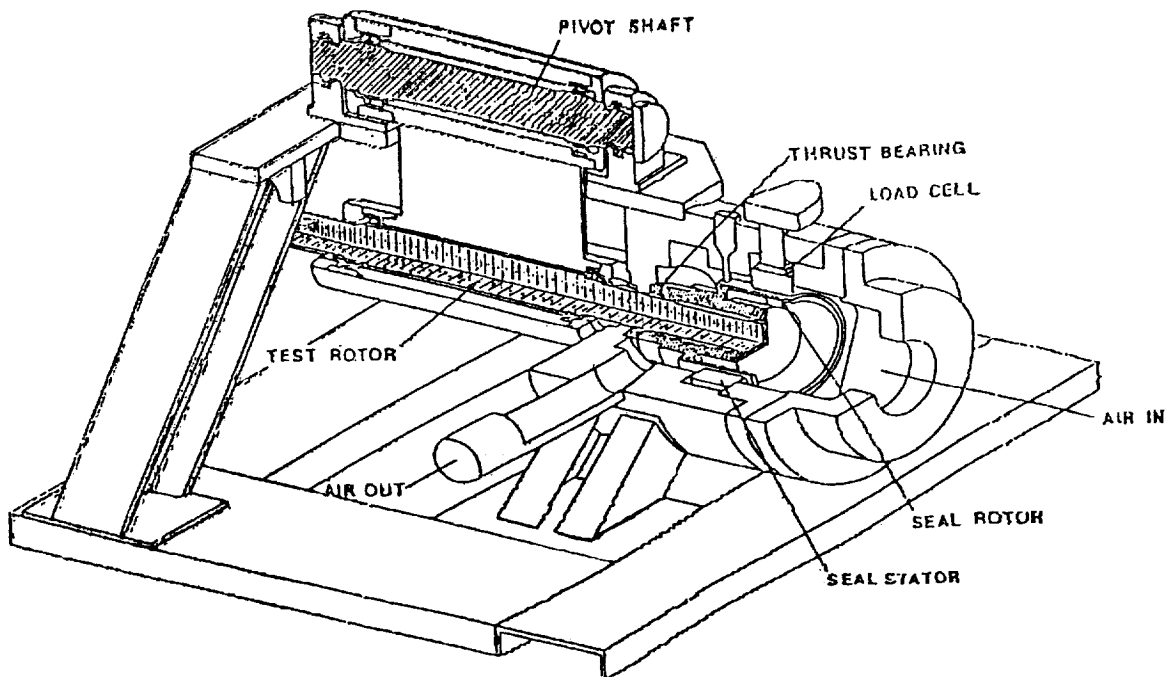
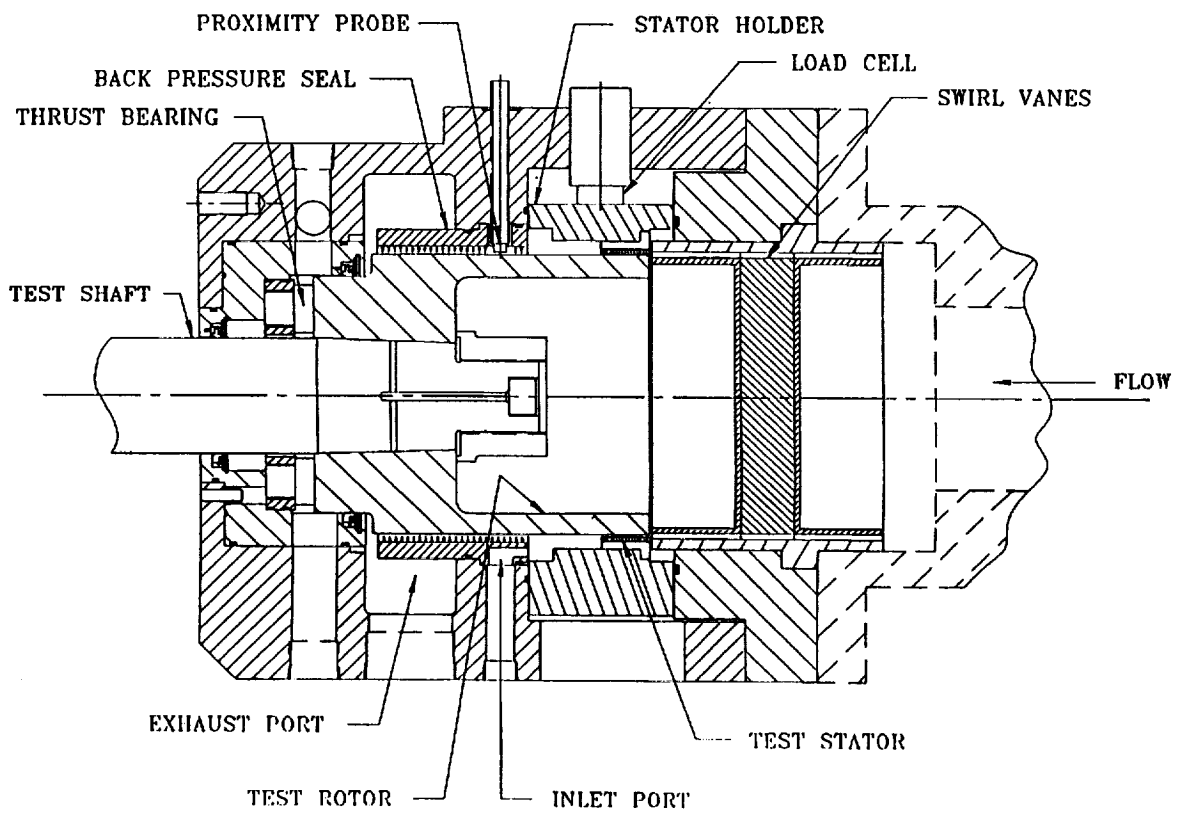
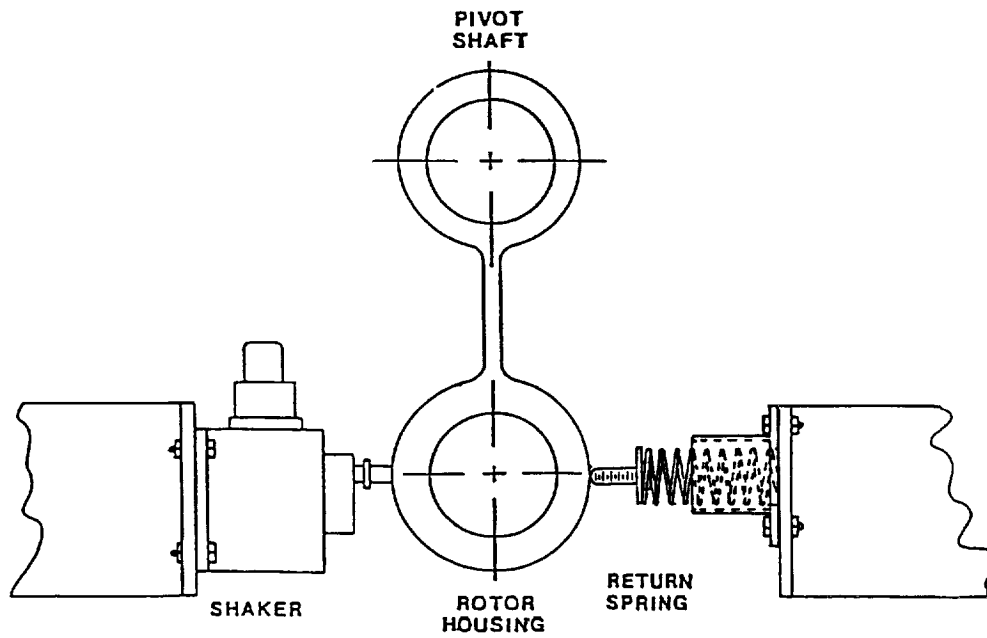


Figure 4. Excitation system.



## Modifications for Coefficient Identification with Eccentric Operation

Identification of all 8 rotordynamic coefficients requires excitation parallel and perpendicular to the static eccentricity vector. The figure below shows the necessary process.

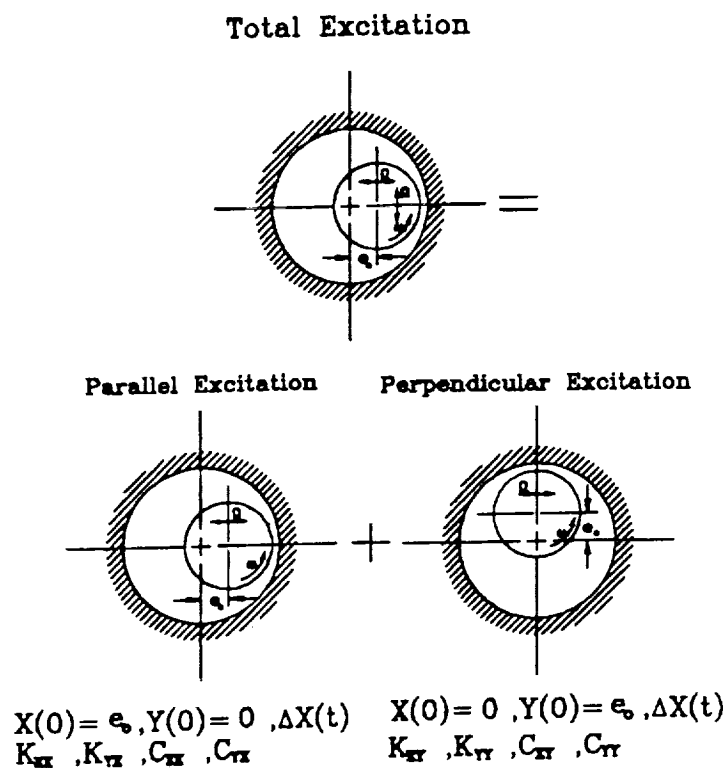


Fig. 6 - Shaking configuration for coefficient identification

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### Test Parameters

The testing apparatus can determine the effects of the following test parameters on the rotordynamic and leakage rates of a seal:

- 1) Rotor Speed
- 2) Inlet Pressure
- 3) Pressure Ratio
- 4) Inlet Fluid Rotation
- 5) Rotor Eccentricity

## PRESENTATION OF EXPERIMENTAL VERSUS THEORETICAL RESULTS

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The independent parameter is ECCENTRICITY. The results presented also show the effects of:

Speed  
Pressure ratio  
Inlet pressure  
Preswirl

| Rotor Speed<br>(rpm)<br>$\omega$ | Inlet Pressure<br>(bars)<br>$P_i$ | Pressure Ratio<br>(-)<br>$P_{ra}$ | Inlet Preswirl<br>in the Direction<br>of Rotation |
|----------------------------------|-----------------------------------|-----------------------------------|---------------------------------------------------|
| 5,000                            | 7.90                              | 0.67                              | None                                              |
| 16,000                           | 11.4                              | 0.55                              | Intermediate                                      |
|                                  | 14.8                              | 0.50                              | High                                              |
|                                  |                                   | 0.45                              |                                                   |

The following coefficients will be discussed:

DIRECT STIFFNESS  
CROSS-COUPLED STIFFNESS  
DIRECT DAMPING  
WHIRL FREQUENCY RATIO

Also discussed will be:

MASS FLOW RATE  
PRESSURE PROFILES

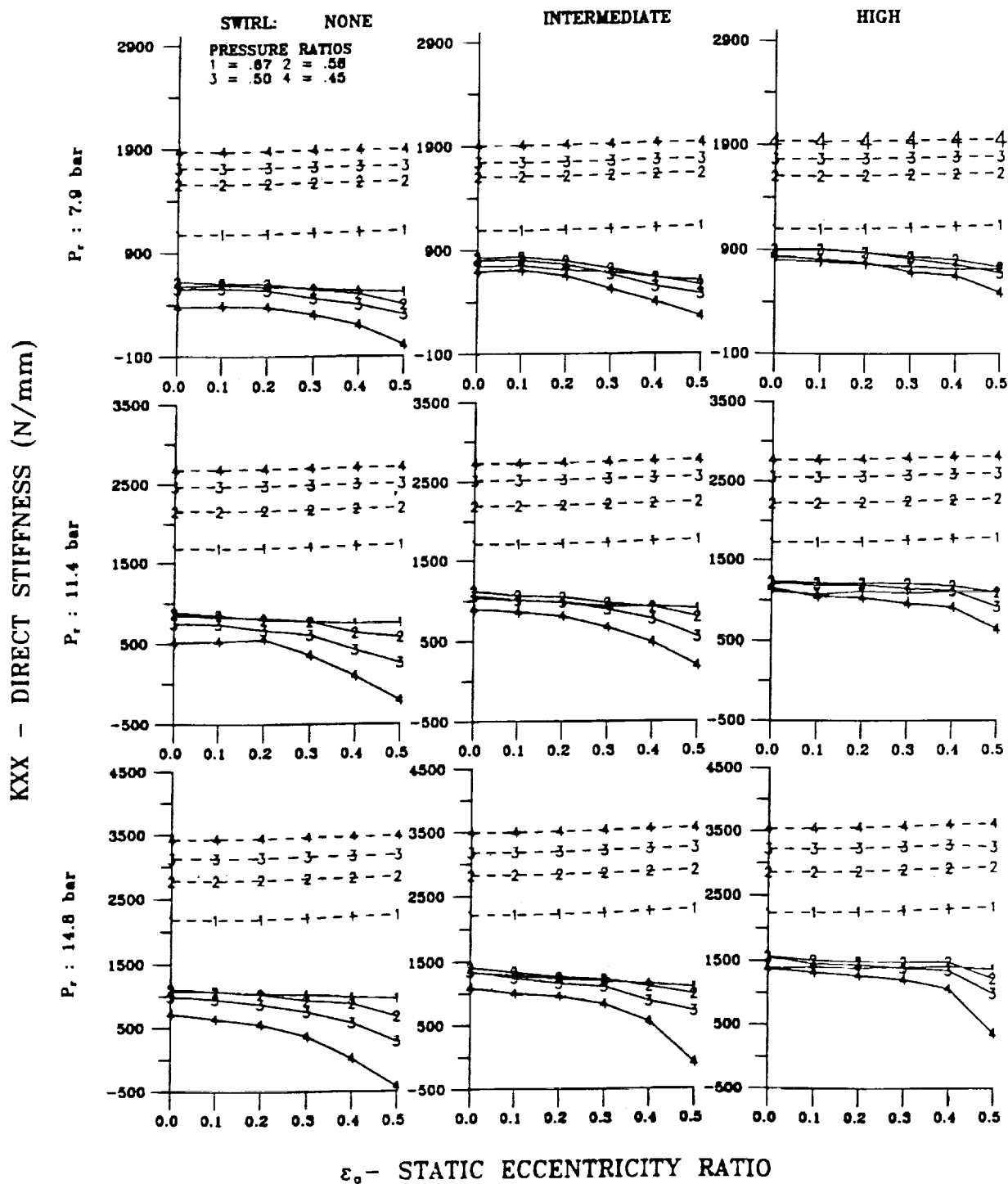


Fig. 9 - Experimental (solid) versus theoretical (dashed) results for direct stiffness,  $K_{XX}$ , as a function of the static eccentricity ratio,  $\epsilon_0$ , for a smooth seal at 16,000 rpm

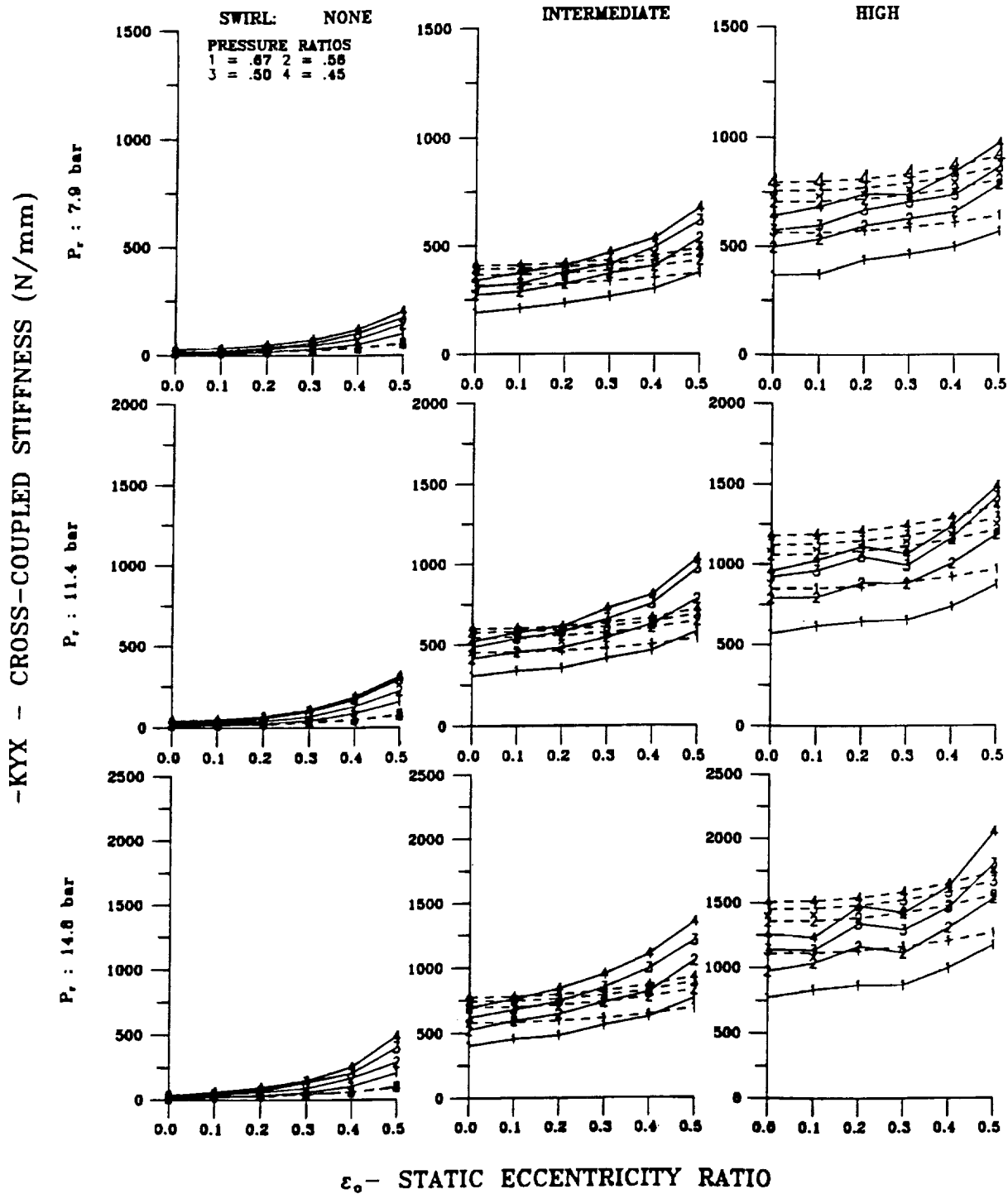


Fig. 10 - Experimental (solid) versus theoretical (dashed) results for cross-coupled stiffness,  $K_{YX}$ , as a function of the static eccentricity ratio,  $\epsilon_o$ , for a smooth seal at 5,000 rpm

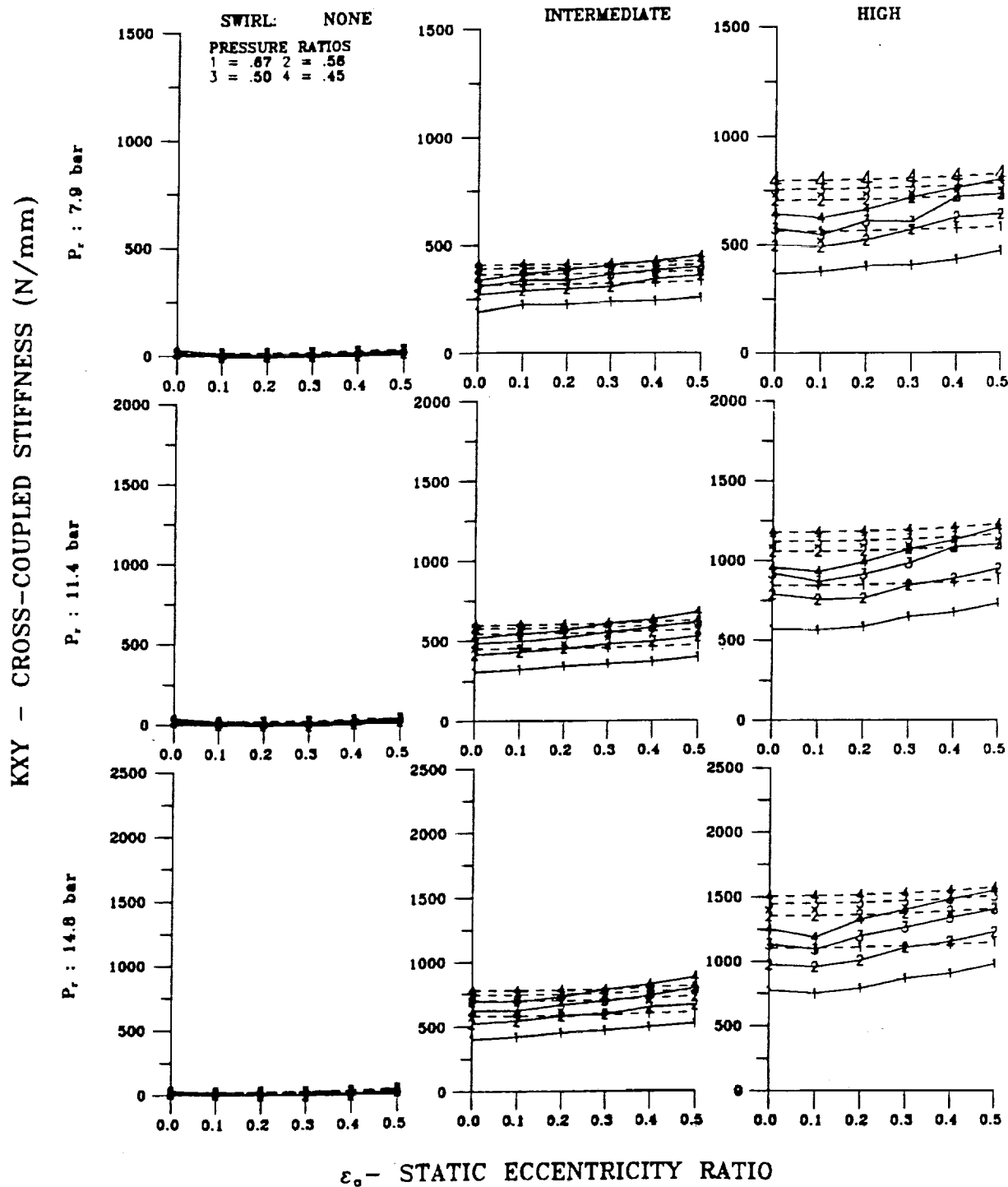


Fig. 11 - Experimental (solid) versus theoretical (dashed) results for cross-coupled stiffness,  $K_{XY}$ , as a function of the static eccentricity ratio,  $\epsilon_o$ , for a smooth seal at 5,000 rpm



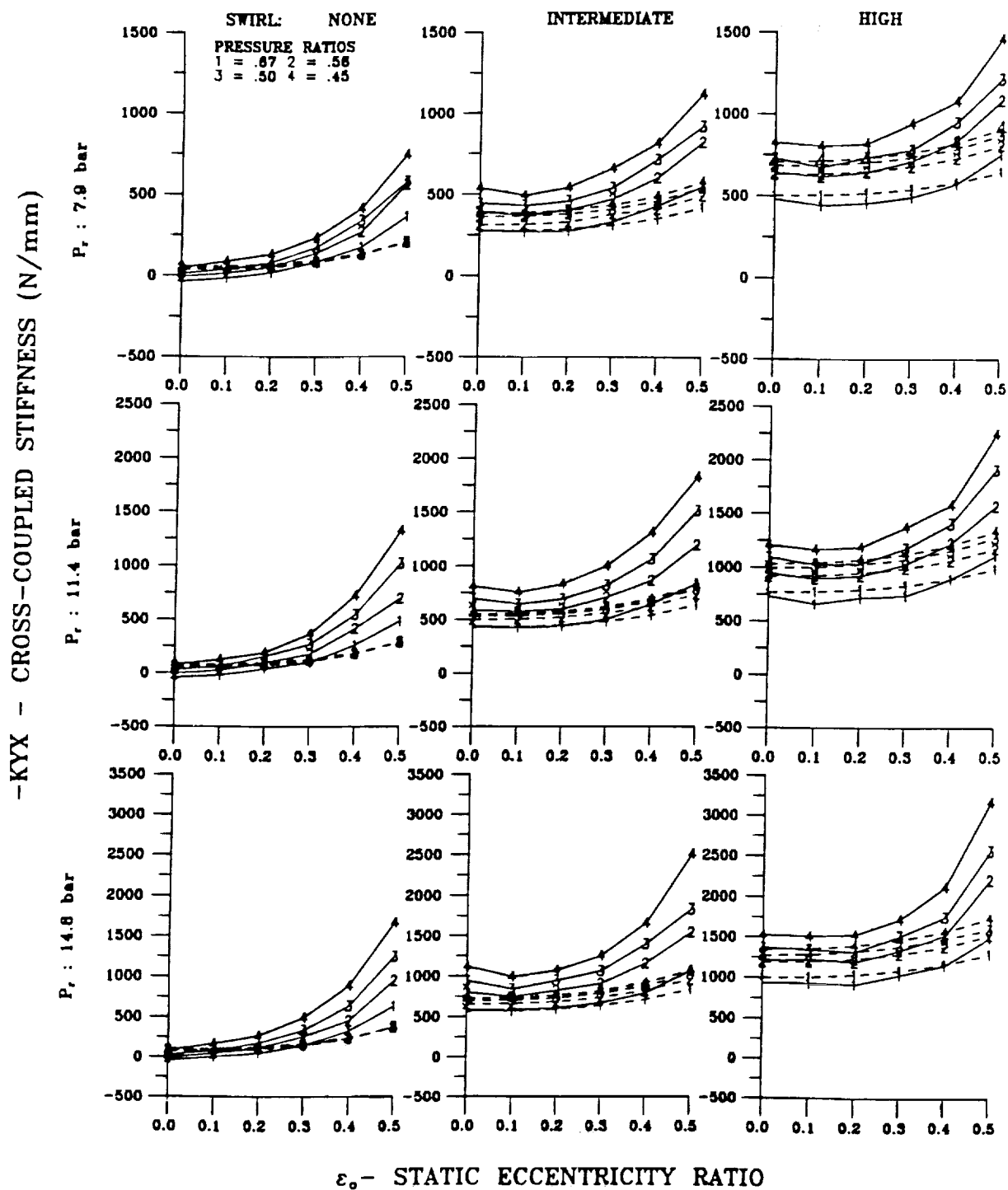


Fig. 12 - Experimental (solid) versus theoretical (dashed) results for cross-coupled stiffness,  $K_{YX}$ , as a function of the static eccentricity ratio,  $\epsilon_o$ , for a smooth seal at 16,000 rpm

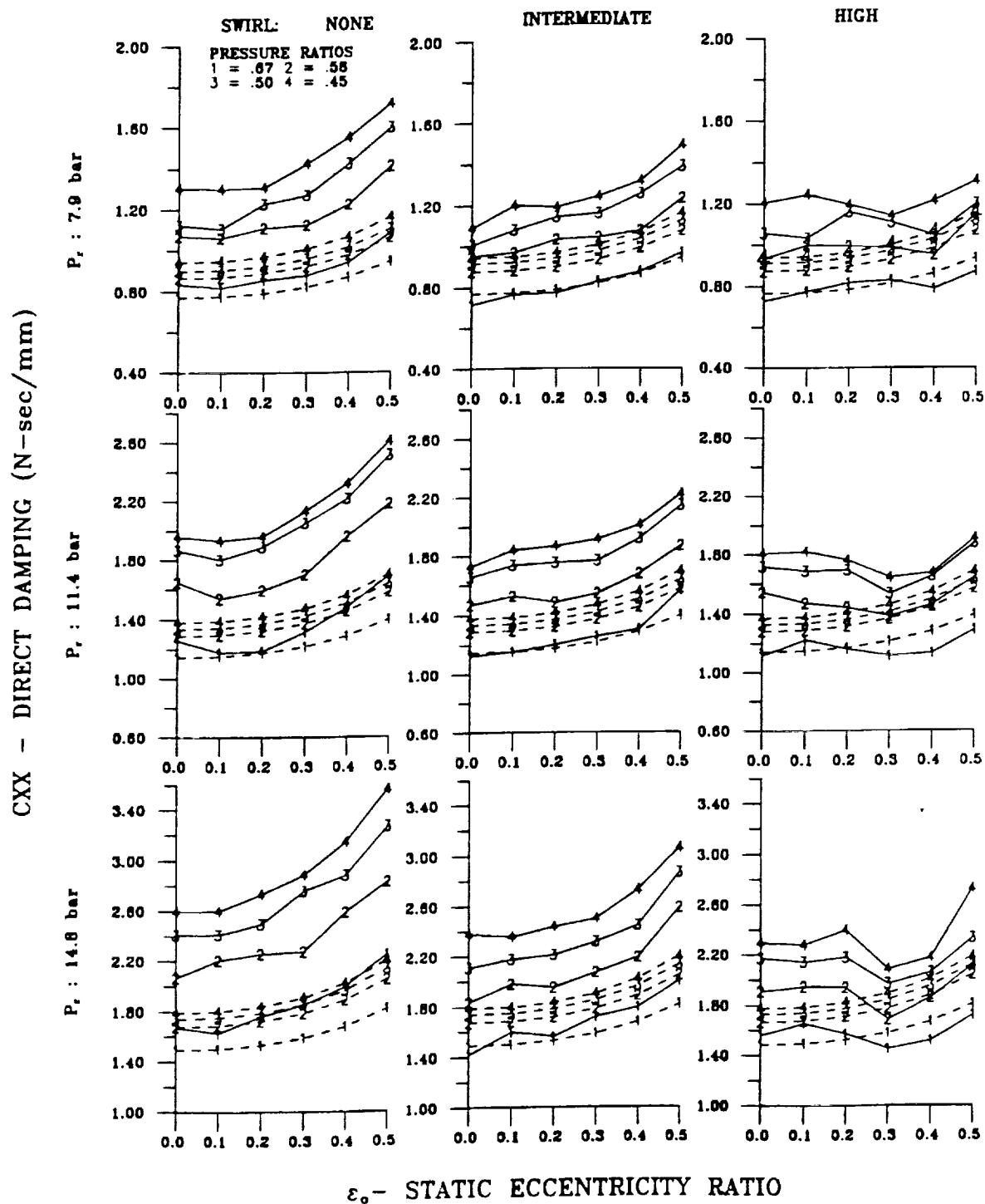


Fig. 13 - Experimental (solid) versus theoretical (dashed) results for direct damping,  $C_{XX}$ , as a function of the static eccentricity ratio,  $\epsilon_0$ , for a smooth seal at 5,000 rpm

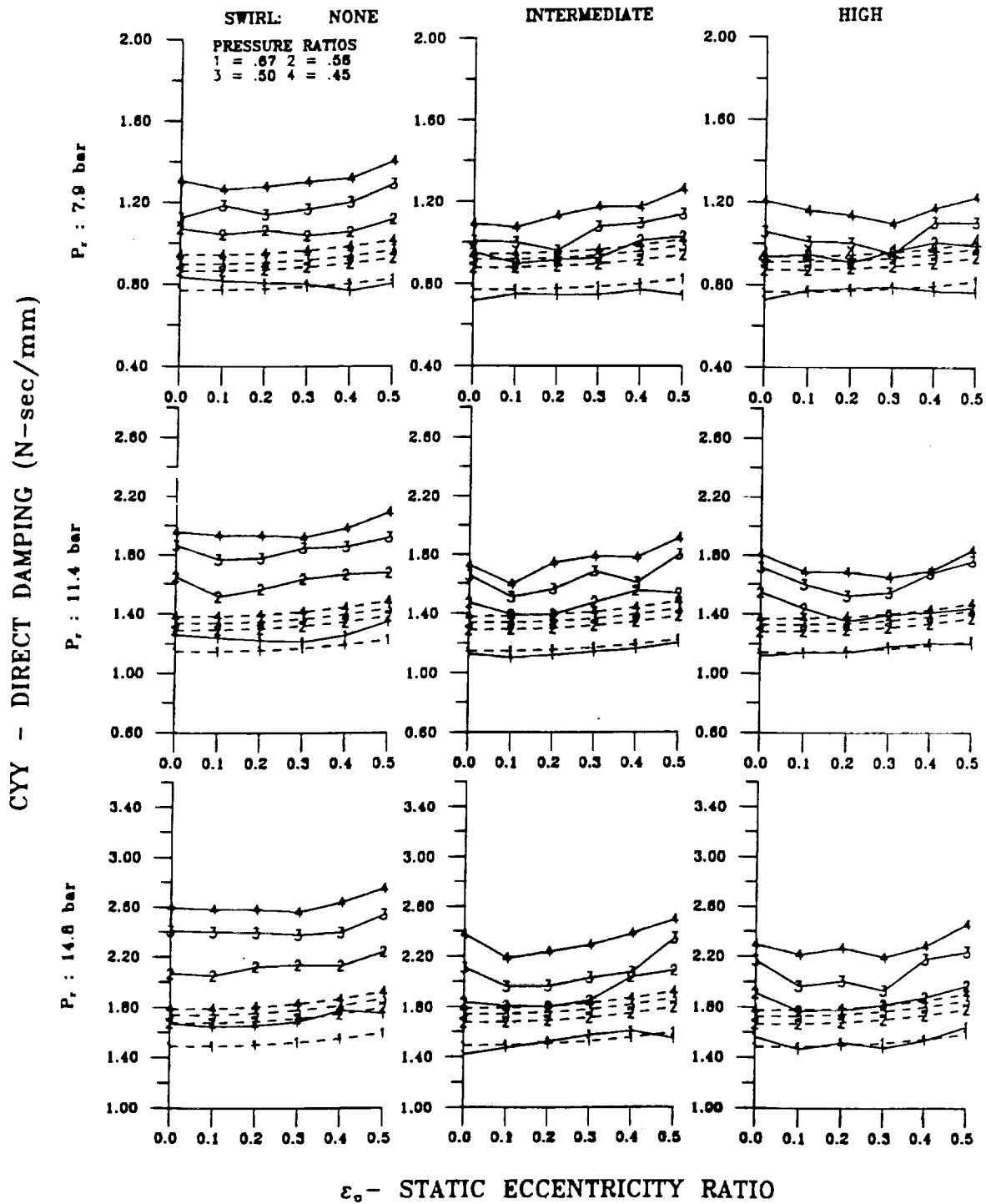


Fig. 14 - Experimental (solid) versus theoretical (dashed) results for direct damping,  $C_{YY}$ , as a function of the static eccentricity ratio,  $\epsilon_o$ , for a smooth seal at 5,000 rpm

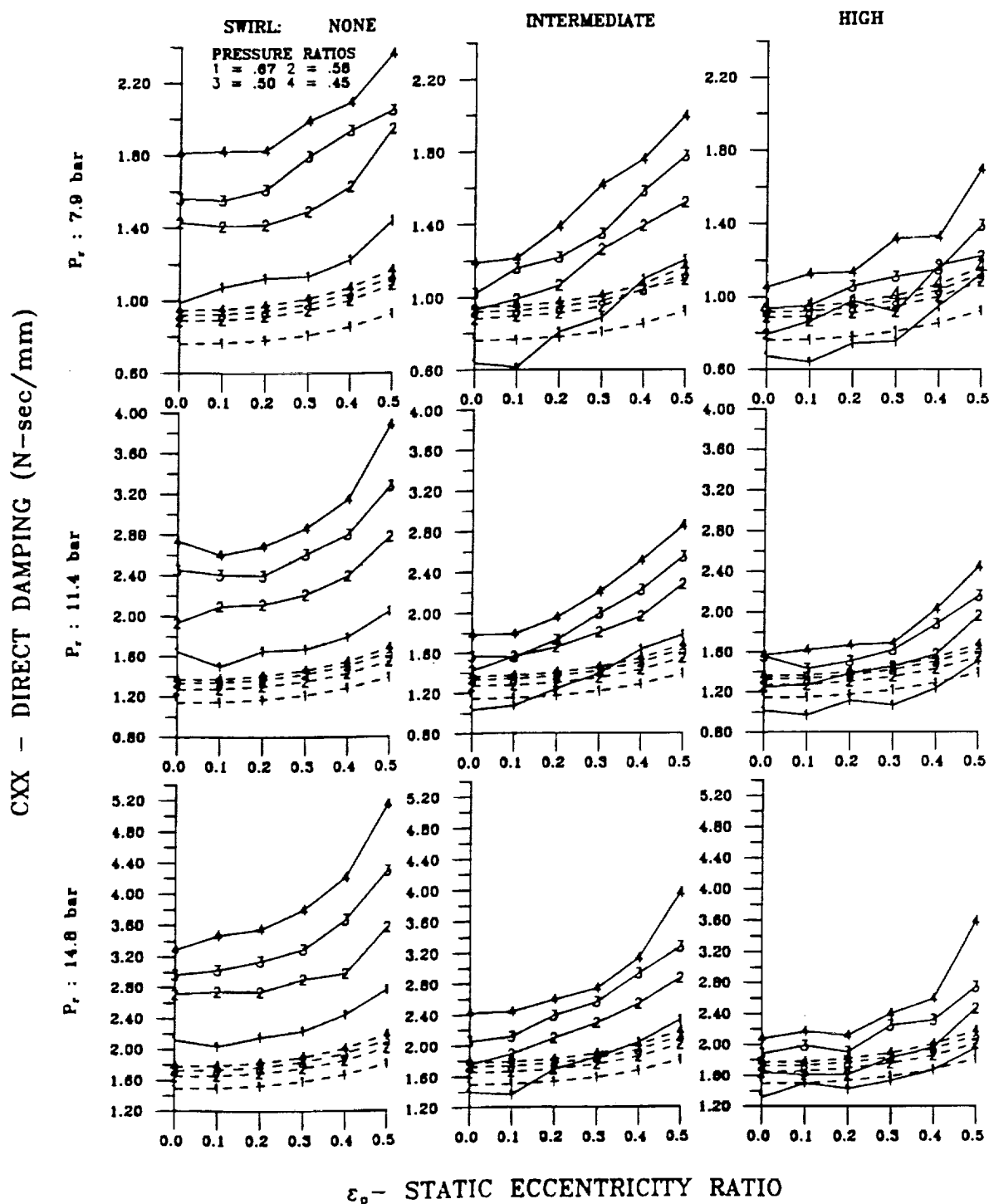


Fig. 15 - Experimental (solid) versus theoretical (dashed) results for direct damping,  $C_{XX}$ , as a function of the static eccentricity ratio,  $\epsilon_0$ , for a smooth seal at 16,000 rpm

## WHIRL FREQUENCY RATIO

The whirl frequency ratio is a means by which to quantitatively determine the rotordynamic stability of a seal. According to Lund (1965), the whirl frequency ratio for eccentric operation is obtained by using the following equations.

$$K_{eq} = \frac{K_{xx}C_{yy} + K_{yy}C_{xx} - C_{yx}K_{xy} - C_{xy}K_{yx}}{C_{xx} + C_{yy}}$$

$$WFR^2 = \frac{(K_{eq} - K_{xx})(K_{eq} - K_{yy}) - K_{xy}K_{yx}}{(C_{xx}C_{yy} - C_{yx}C_{xy})\omega^2}$$

The onset speed of instability is defined by the following equation,

$$\omega = \frac{\omega_{n1}}{\Omega_w}$$

where  $\omega_{n1}$  is the first critical speed of the rotor and  $\Omega_w$  is the whirl frequency ratio.

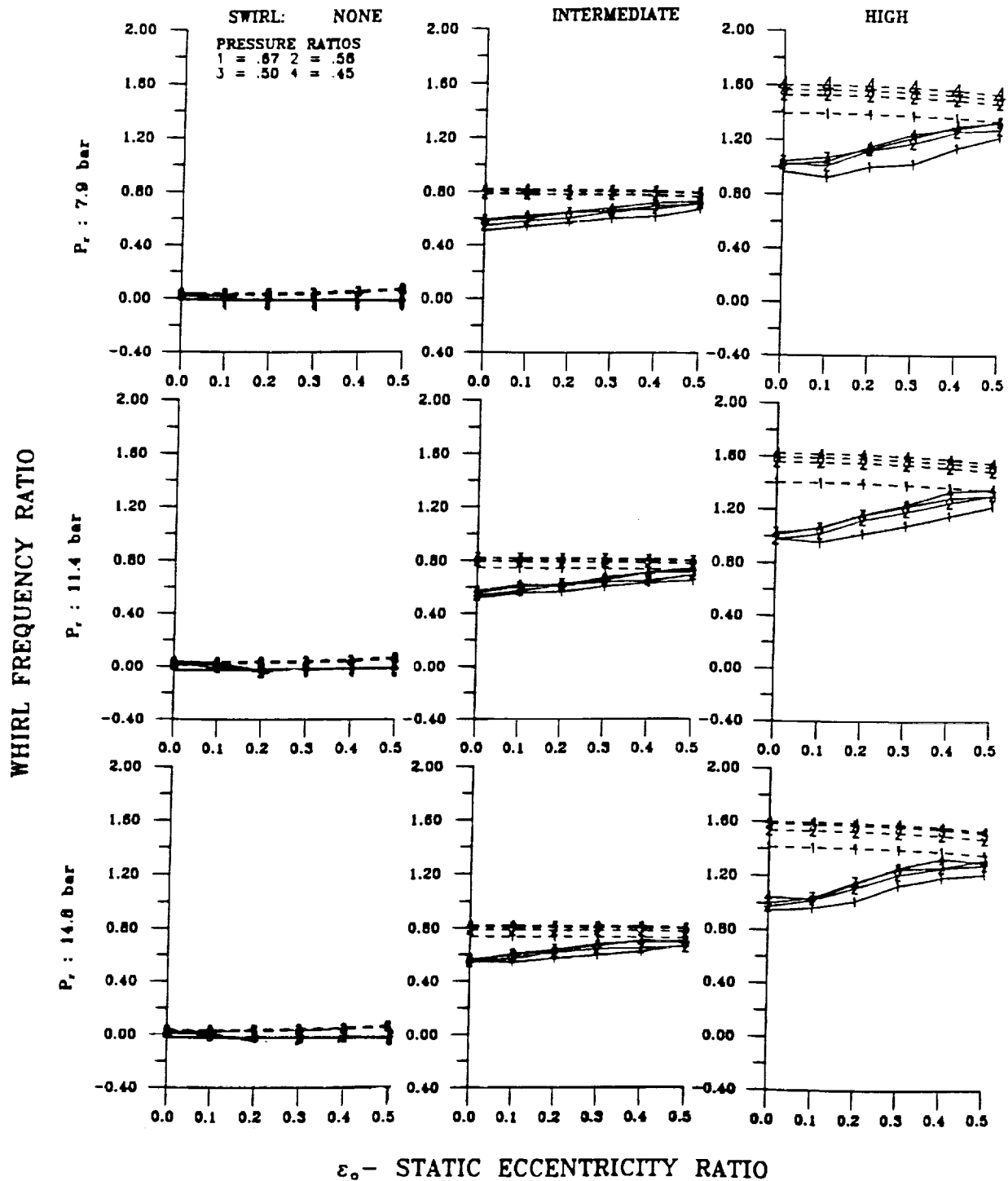


Fig. 16 - Experimental (solid) versus theoretical (dashed) results for the whirl frequency ratio as a function of the static eccentricity ratio,  $\epsilon_o$ , for a smooth seal at 5,000 rpm

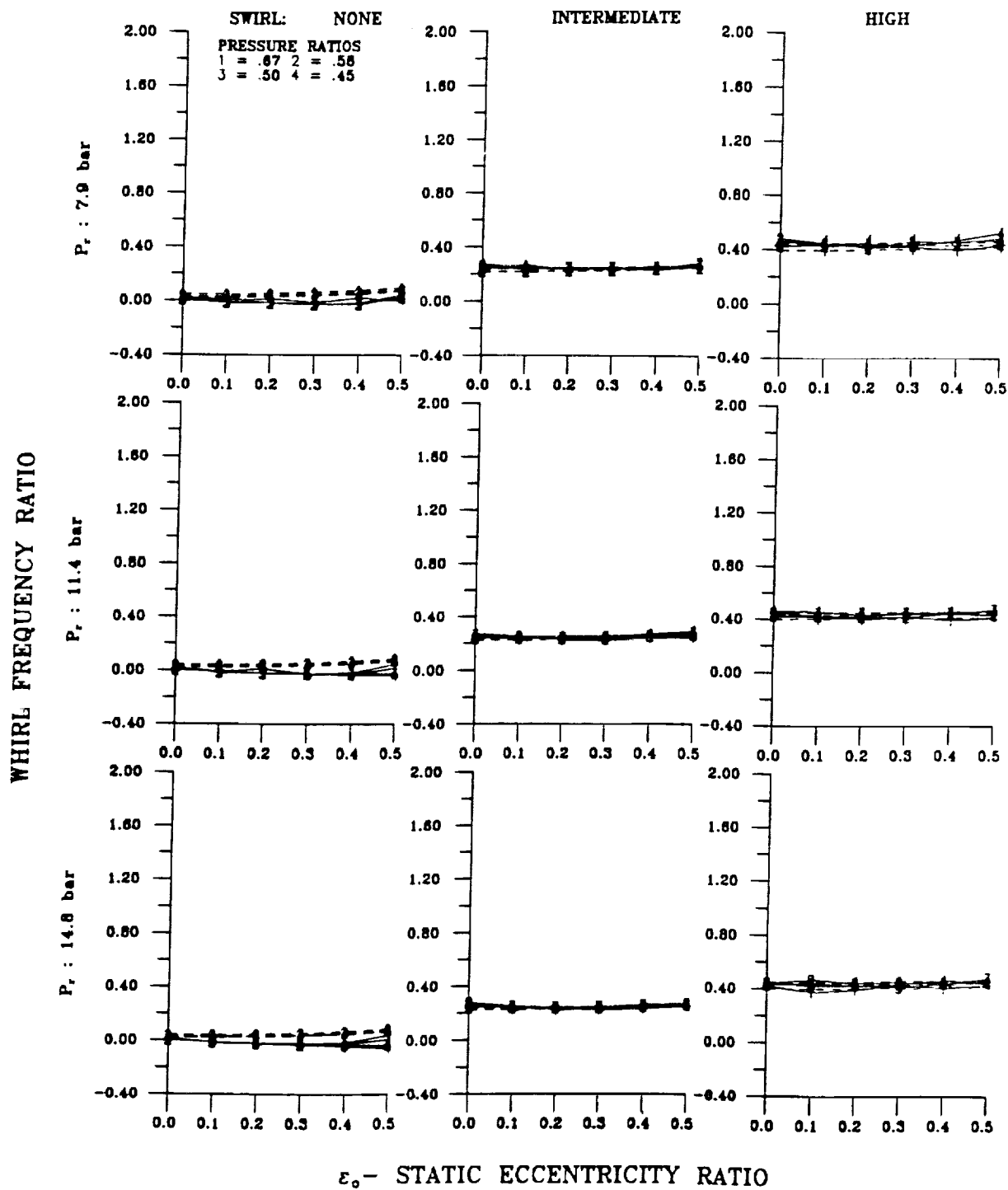
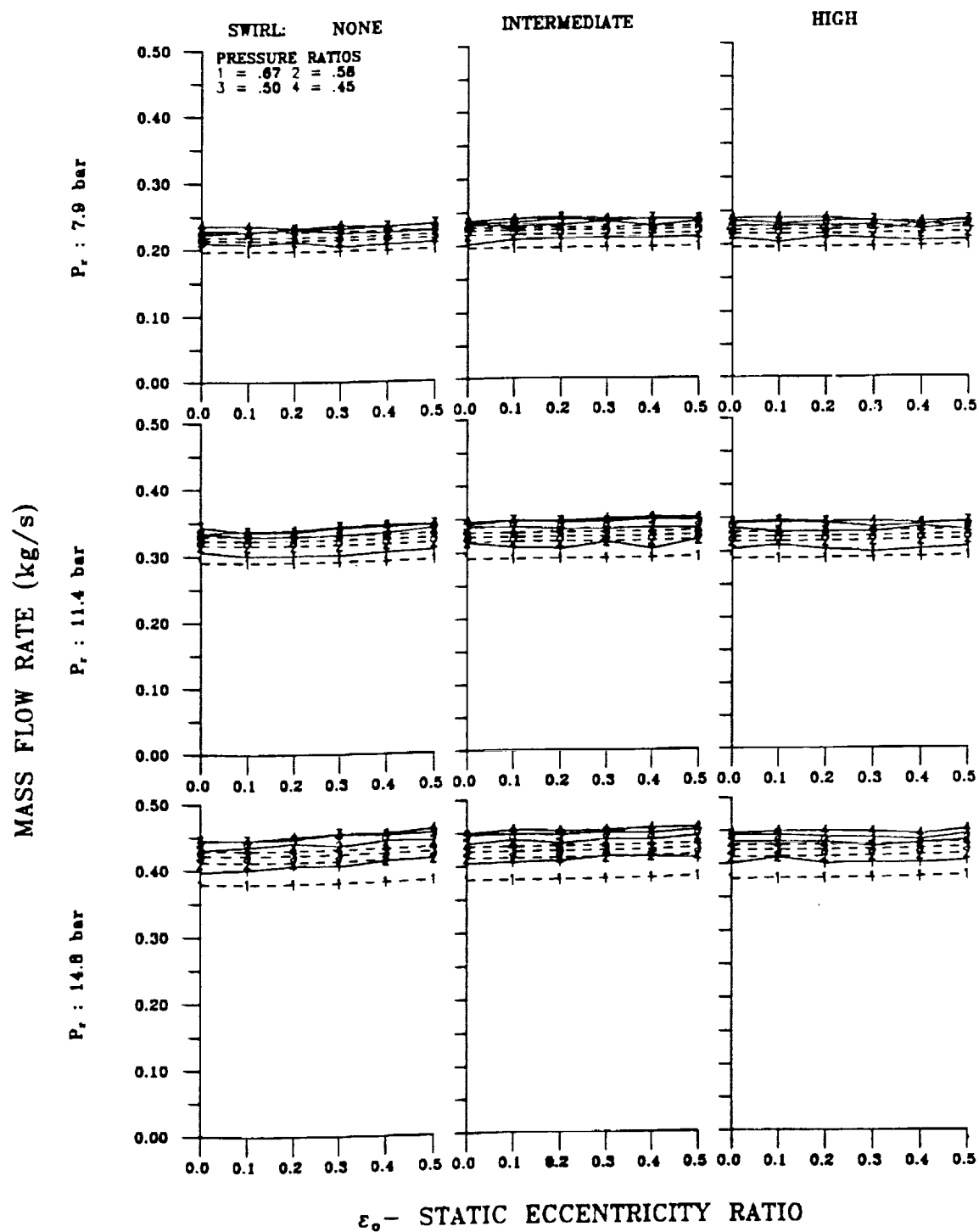


Fig. 17 - Experimental (solid) versus theoretical (dashed) results for the whirl frequency ratio as a function of the static eccentricity ratio,  $\epsilon_o$ , for a smooth seal at 16,000 rpm



**Fig. 18** - Experimental (solid) versus theoretical (dashed) results for mass flow rate as a function of the static eccentricity ratio,  $\epsilon_0$ , for a smooth seal at 5,000 rpm



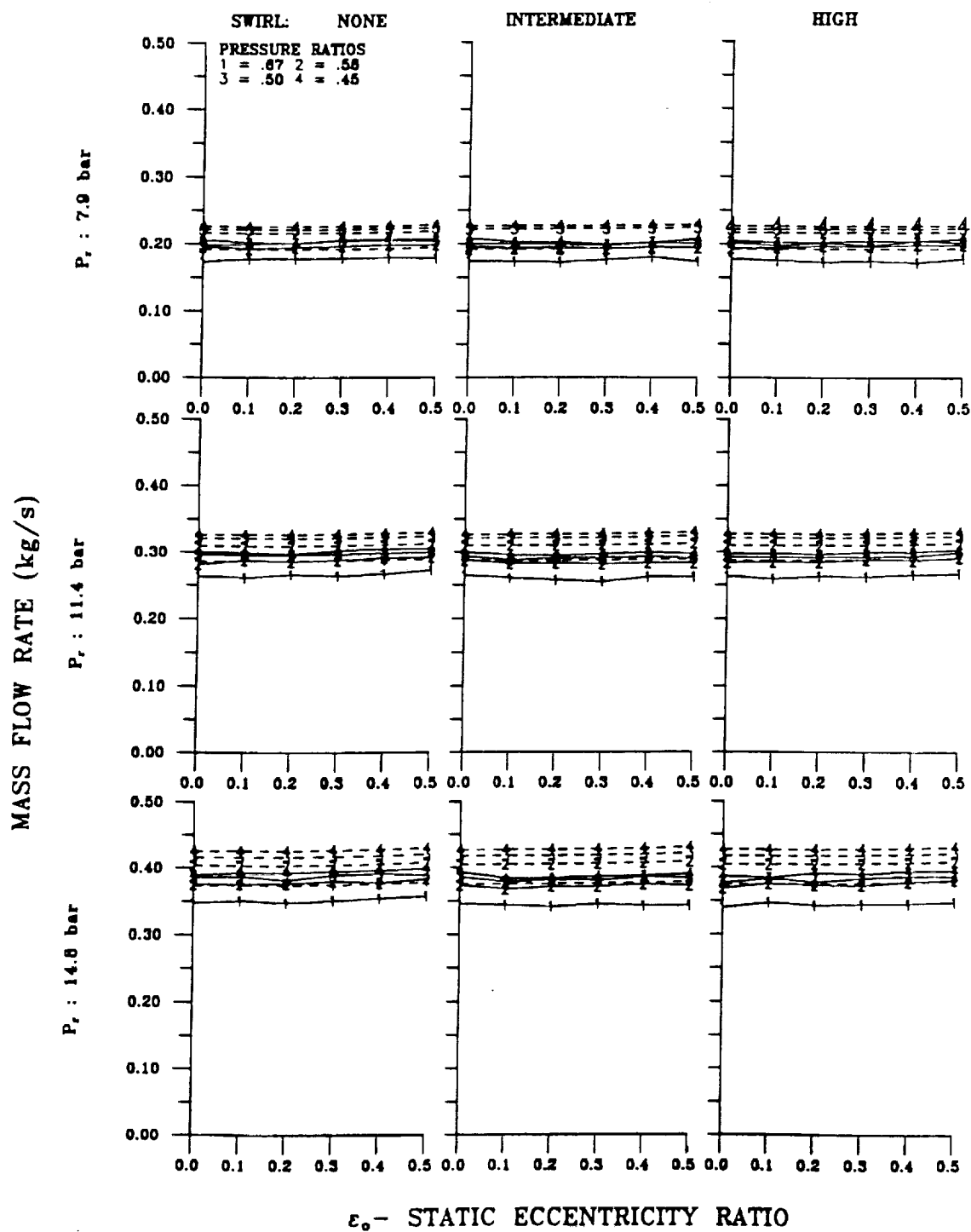


Fig. 19 - Experimental (solid) versus theoretical (dashed) results for mass flow rate as a function of the static eccentricity ratio,  $\epsilon_0$ , for a smooth seal at 16,000 rpm

## CONCLUSIONS

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### Experimental Results

- Direct stiffness,  $K_{xx}$ , decreases with increasing eccentricity, inlet pressures, and slightly with increasing inlet preswirl
- Cross-coupled stiffness,  $K_{yx}$ , increases with increasing eccentricity, preswirl and speed
- Direct damping,  $C_{xx}$ , increases with increasing eccentricity, speed, and inlet preswirl
- At 5,000 rpm and no swirl the whirl frequency is basically zero for all eccentric positions. For the configurations with swirl the whirl frequency values increase with increasing eccentricity. At 16,000 rpm the whirl frequency ratio does not change with eccentricity.
- At 5,000 and 16,000 rpm the leakage is invariant to changes in the static eccentricity ratio
- Linear pressure profiles exist for all eccentric positions

### Theoretical versus Experimental Results

- The analytical results overpredict the experimental results for the direct stiffness values and incorrectly predict increasing stiffness with decreasing pressure ratios
- Theory correctly predicts increasing cross-coupled stiffness,  $K_{yx}$ , with increasing eccentricity and inlet preswirl
- Direct damping,  $C_{xx}$ , underpredicts the experimental results, but the analytical results do correctly show that damping increases with increasing eccentricity
- The whirl frequency values predicted by theory are insensitive to changes in the static eccentricity ratio. Although these values match perfectly with the experimental results at 16,000 rpm, the results at the lower speed do not correspond
- Theoretical and experimental mass flow rates match at 5,000 rpm, but at 16,000 rpm the theoretical results overpredict the experimental mass flow rates
- Theory correctly shows the linear pressure profiles and the associated entrance losses with the specified rotor positions